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EFFECTS OF INGESTION ON THE FLOW AND HEAT TRANSFER IN A ROTOR-STATOR SYSTEM

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ABSTRACT

Ingestion (or ingress) of hot mainstream gas into the wheel-spaces between the rotor and stator discs in the high pressure stages of gas turbines is one of the most important internal cooling problems for engine designers. A rim seal at the periphery of the wheel-space and a radial outflow of sealing air are used to reduce or prevent this damaging ingestion. The sealing air is also used for rotor disc cooling. In this paper, a simplified axisymmetric computational fluid dynamics (CFD) model of a gas turbine rotor-stator wheel-space is used to predict the effects of ingress on the flow and heat transfer. The steady flow computations are carried out using the CFD code ANSYS/CFX and the shear stress transport (SST) turbulence model. The rotational Reynolds number, inlet flow rates and thermal boundary settings are selected to represent the operating conditions of an ingress experimental rig, and comparisons are made between computations and measurements of heat transfer in the wheel-space. The computed heat transfer from the steady model (in terms of values of local Nusselt number on the rotor) is in broad agreement with measurements obtained using thermochromic liquid crystal in a transient experiment. Alternative wall-surface thermal boundary conditions are investigated in order to study the sensitivity of computed Nusselt numbers to these modelling assumptions.

KEY WORDS: Convection, Computational methods, Cooling turbine blade. Gas turbine cooling. Rotating flow.

1. INTRODUCTION

In gas turbines, some of the air from the compressor does not enter the combustion chamber but is used instead for turbine component cooling and for sealing. In modern gas turbine engines about 20% of the compressed air is used for this internal cooling system. The use of compressor air for cooling purposes reduces engine efficiency; hence an effective cooling system is essential in optimising engine performance. A rim seal is used to prevent or reduce the ingestion of hot mainstream gas from the turbine annulus into the wheel-spaces between rotor and stator discs. As illustrated in Fig. 1 (taken from Pountney et al [1]), the sealing air supplied from the compressor is fed into the wheel-space and together with the rim seal design can be used to control the amount of ingestion (referred as ingress). The use of too little sealing air can cause overheating and fatigue and reduce the operating life of the discs, however too much sealing air impacts adversely on engine efficiency.

As described by Owen et al [2,3], externally-induced (EI) ingress of hot gas into the wheel-space occurs when the non-axisymmetric pressure distribution created by the flow past the stationary vanes and rotating blades in the turbine annulus causes ingress and egress to occur in regions of the rim seal circumferentially where the pressure in the annulus is higher and lower respectively than the corresponding pressure in the wheel-space. On the other hand, even if the external distribution of pressure were axisymmetric, rotationally-induced (RI) ingress can still occur. The rotation of the fluid core in the wheel-space (Fig. 1) creates a radial gradient of pressure so that the pressure can fall below that outside, leading to ingress of fluid through the rim seal into the wheel-space, where it is drawn into the boundary layer flowing inward on the stator.

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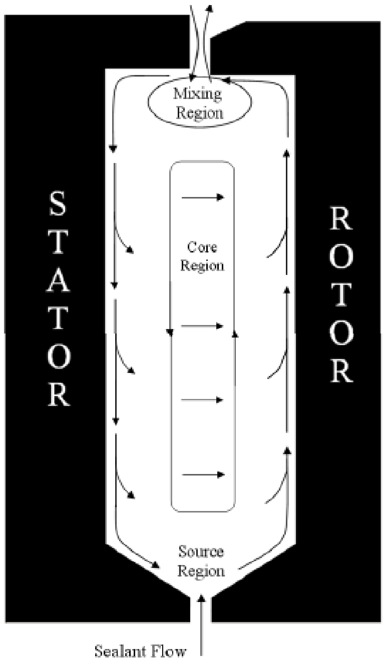

Fig. 1 Flow structure in a rotor-stator wheel-space (from Pountney et al [1])

The aim of this paper is to use an axisymmetric computational fluid dynamics (CFD) model to predict the heat transfer in a rotor-stator wheel-space including the effects of externally-induced ingress. The model provides information about the fluid dynamics and heat transfer in the wheel-space that could be useful for engine designers. While the axisymmetric model is a great simplification compared to the complex, and costly, 3D unsteady CFD codes often used in gas-turbine research and development, engine designers also require more time and cost-effective tools to rapidly test engine seal components and cooling technologies.

Due to the importance of ingress, there have been a large number of investigations relating to this problem involving experiments, computations and theoretical work. Concentration measurements are widely used in experimental studies. These use a convenient tracer gas, such as CO_2, introduced into the sealing flow n order to measure the amount of ingestion into the wheel-space and quantify the sealing effectiveness, ε, non-dimensionally, so that $\varepsilon = 0$ when there is no sealing and $\varepsilon = 1$ when the wheel-space is fully sealed from ingestion. CFD has been used to compute turbine annulus pressure distributions, wheel-space sealing effectiveness and the fluid dynamics inside and around the rim seal region.

Bayley and Owen [4] used a simple rotor-stator rig with an axial-clearance rim seal and without an external annulus to carry out ingress experiments. They showed that increasing the sealing flow rate increases the pressure in the wheel-space, consequently reducing ingestion. The minimum value of non-dimensional sealing flow rate $C_{w,o}$ ($= \dot{m}_o / \mu b$) needed to prevent ingress depended on the seal clearance ratio, G_c,($= s_c/b$) and the rotational Reynolds number, Re_ϕ ($= \Omega b^2/\mu$), but was little affected by the rotor-stator clearance, s.

Graber et al. [5] used concentration measurements for several rim seal geometries with no circumferential variation of the external pressure. They plotted sealing effectiveness against non-dimensional sealing flow rate and identified the effects of G_c and Re_ϕ, finding no effects of external swirl on effectiveness.

Abe et al. [6] used a turbine rig in experiments with vanes in the external annulus upstream of the rim seal in order to show that the external flow in the annulus can also cause ingress. The authors identified three factors that affected ingress: the ratio of the velocities of the sealing air and the flow in the annulus; the rim seal clearance; the shape of the rim seal.

Roy et al. [7] described experimental measurements for a single stage axial-flow turbine with two different vane-blade configurations. The main difference between the configurations was the inlet vane turning angle. The measurements suggested that the instantaneous pressure field in the annulus is pivotal to ingestion. The 3D and unsteady pressure field is influenced by the mainstream flow rate, rotor speed, and sealing flow rate.

Zhou et al. [8] described experiments and computations for three wheel-space configurations having different aspect ratios. At low sealing rate, they also identified the regions of ingress and egress around the rim seal through PIV images of instantaneous velocity. For the computations, simulations for a 14.4° circumferential sector model under-predicted the measured ingestion into the wheel-space. It was suggested that the wheel-space sector model could not predict the rotating low-pressure regions that augment ingestion.

Mirzamoghadam et al. [9] reported 3D CFD computations for a full stage HP turbine rotor-stator wheel-space. Based on these computational results, they found that the asymmetrical annulus pressure can give rise to ingestion even at relatively high sealing flow rates. Ingestion mixing efficiency was defined using temperature data as:

$$\eta_{ing} = (T_r - T_c)/(T_h - T_c) \tag{1}$$

T_c, T_h and T_r are coolant inlet temperature, annulus temperature and local temperature at radius r inside the wheel-space respectively. With this definition, mixing efficiency equal to zero implies that no ingress occurs.

Laskowski et al. [10] found significant differences in the distribution of effectiveness (hence the patterns of ingestion) in the rim seal region between steady and unsteady simulations of ingress. Values of sealing effectiveness based on temperature were found to be lower than those based on concentration, due to the increase in temperature in the wheel-space through windage (frictional heating).

Teuber et al. [11] carried out unsteady RANS computations to investigate the fluid mechanics in a 3D model of a turbine stage. Computed radial variations of swirl ratio β (where β is the ratio of the angular velocity of the fluid at any location to the angular velocity of the rotor) in the rotating core of fluid in the wheel-space (between the boundary layers on the rotor and stator discs, see Fig. 1) showed good agreement with measured values. The authors demonstrated that the minimum sealing flow required to prevent ingestion increases as the annulus Mach number increases.

O'Mahoney et al. [12] presented Large-Eddy Simulations (LES) results for turbine rim seal ingestion that were in better agreement with published experimental data than results from a corresponding unsteady RANS simulation. Julien et al. [13] found that large scale flow structures occurred in the wheel-space in simulations at low sealing flow rates, and that the associated circumferential pressure variations in the wheel-space reduced sealing effectiveness. Cao et al. [14] and Jakoby et al. [15] also studied instabilities in the wheel-space caused by ingestion and the effect of circumferential sector modelling simplifications. Rabs et al. [16] found that a full 360° domain model gave almost no improvement, when compared to sector models, if no large scale rotating structures occur. The use of sector models was recommended due to the significant savings in computational cost.

Wang and Wilson [17] simplified the physical situation drastically by using an axisymmetric CFD model of a rotor-stator wheel-space and prescribing ingestion into the wheel-space through boundary conditions applied at the location of the rim seal, in order to study qualitatively the effect of ingress on the flow and heat transfer in the wheel-space. In prior validation of the wheel-space model, good agreement was obtained between computations and measurements of flow and heat transfer made in a different rotor-stator system without ingress, when the sealing flow rate was sufficiently high. For computations involving ingress the ingestion boundary conditions specified were approximate, however reasonable agreement was obtained between computed and measured values of swirl ratio. In the present paper, the same "prescribed ingestion" model is used to make direct comparisons between computed values of Nusselt number on the rotor and measurements made by Pountney et al [1] for different levels of ingestion. Such comparisons have not been reported previously.

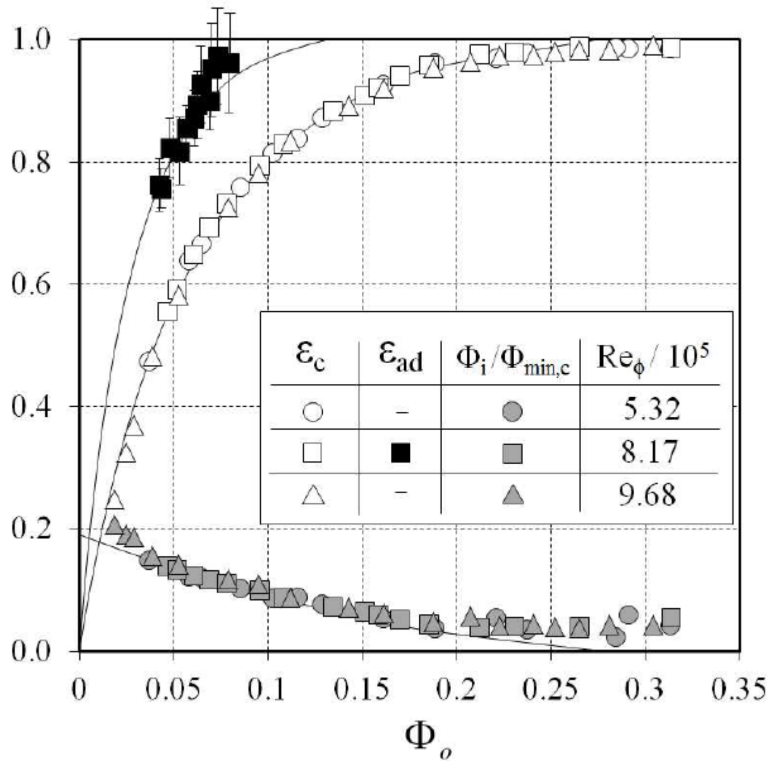

Fig. 2 Variation of concentration-based effectiveness on the stator, thermally-based adiabatic effectiveness on the rotor and ingress flow parameter with non-dimensional sealing flow parameter Φ_o (from Pountney et al [1]). Lines – theoretical orifice model results [1]

2. EXPERIMENTAL AND THEORETICAL STUDIES OF INGRESS AT THE UNIVERSITY OF BATH

Owen et al. [2-3] developed orifice models for the ingress phenomenon. Although the equations are derived for inviscid flow, discharge coefficients (for both ingress and egress fluid streams) are introduced to account for losses, and the theoretical equations for the rim seal effectiveness ε are obtained as a function of a non-dimensional sealing flow parameter Φ_o ($=C_{w,o}/2\pi G_c Re_\phi$). For externally induced ingress [2]:

$$\frac{\Phi_o}{\Phi_{mn,EI}} = \frac{\varepsilon}{\left[1+\Gamma_c^{-2/3}(1-\varepsilon)^{2/3}\right]^{3/2}} \tag{2}$$

where Γ_c, the ratio of the discharge coefficients for ingress and egress, is an empirical constant which is different for different rim seal geometries. $\Phi_{imin,EI}$ is the minimum value of the sealing flow parameter (hence sealing flow rate) required to prevent externally-induced ingress.

Sangan et al. [18 - 20] carried out experiments in order to measure concentration-based sealing effectiveness, ε_c, and hence deduce Γ_c for different rim seal geometries. Pountney et al [1] carried out similar experiments, and Fig. 2 shows a comparison between experimental measurements of ε_c and the theoretical variation of (at a fixed monitor location on the stator inside the wheel-space) for an axial-clearance seal according to the above equation. At the “design point” in these experiments, the flow rate in the annulus (at a fixed value of Re_ϕ) was kept constant and the sealing flow rate was varied in order to vary Φ_o. Fig. 2 shows that the fit between the orifice model theoretical result from Eq. (2) and the measured variation of ε_c with Φ_o, optimised using a maximum likelihood statistical technique as described by Zhou et al. [21], is very good. (The results in Fig. 2 indicate that $\Phi_{imin,EI} \approx 0.3$ for this seal geometry.).

Pountney et al [1] also made heat transfer measurements on the rotor surface for the same experimental rig using heated sealing flow and thermochromic liquid crystal in a transient experiment, and results for a thermally-based adiabatic sealing effectiveness, ε_{ad}, are also shown in Fig. 2. This adiabatic sealing effectiveness for the rotor is higher than the concentration-based effectiveness measured on the stator at the same radius. This was attributed to the "buffering" effect of the sealing flow that is entrained into the boundary layer flowing radially outward on the rotor (see Fig. 1).

Fig. 2 also shows the variation with Φ_o of the non-dimensional *ingress* flow parameter Φ_ι (in terms of the ratio $\Phi_\iota/\Phi_{min,EI}$) which can be derived from the measured effectiveness (Sangan et al [18]). In the present computations, these values of Φ_ι are used to prescribe the ingestion mas flow rate as a boundary condition.

3. COMPUTATIONAL MODEL

The simplified computational model of the University of Bath ingress experimental rig is that described and used by Wang and Wilson [17] for a study that included validation of the turbulence model by comparison with measurements for flow and heat transfer in a rotor-stator system with a radial outflow of sealing air. The axisymmetric geometry, model and the mesh used are illustrated in Fig. 3. Following sensitivity tests, the mesh contained 16,000 cells. Computations were carried out using ANSYS/CFX Version 11 and the Shear-Stress Transport (SST) turbulence model. Normalised residual tolerances of 2×10^{-7} were achieved at convergence for the conservation of mass, momentum (through the velocity components), energy and turbulence quantities. A low-Reynolds number turbulence modelling approach was used with maximum y+ values of around 1.5. Blended upwind differencing was used for advection. Cyclic symmetry conditions are applied at the circumferential faces of the 0.2° sector model.

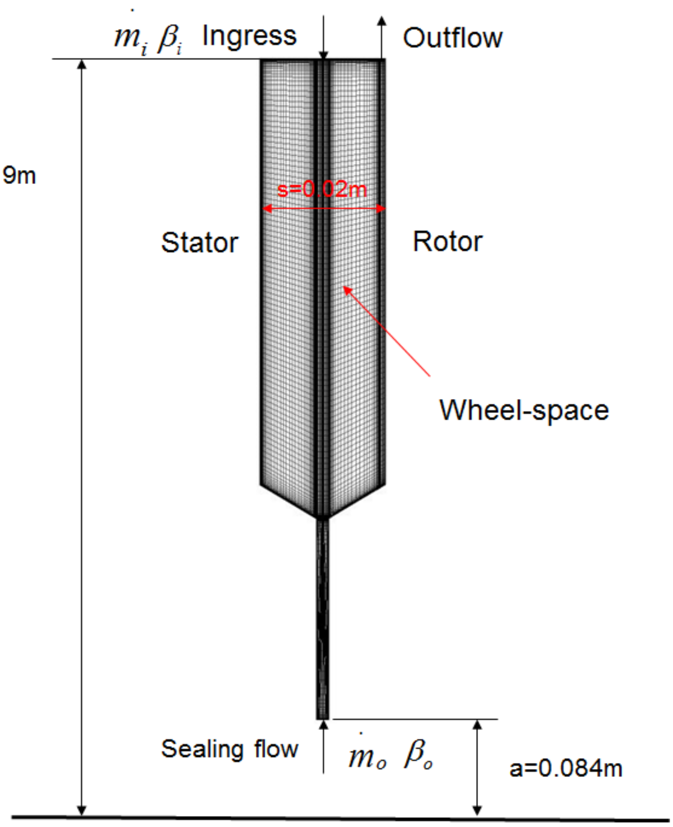

Fig. 3 Wheel-space geometry and computational model

Sealing flow enters at the inlet boundary at inner radius a = 0.117m and flow leaves through the rotor side outlet at the outer radius b = 0.19m (a/b = 0.62). The wheel-space gap ratio is G (=s/b) = 0.1 and the seal clearance gap ratio is $G_c = 0.01$. Uniform values of velocity components and temperature are prescribed at the inlet at r/b = 0.62, giving zero swirl ratio β at inlet. For the outlet boundary, an average static pressure condition is set. A fixed uniform rotor temperature is used as described below, with other surfaces being assumed to be adiabatic.

Flow and heat transfer computations are carried out at a fixed rotational Reynolds number $Re_\phi = 8.17 \times 10^5$ for different values of sealing flow rate. By coincidence, for the combination of geometric and test conditions considered here, $\Phi_o \approx \lambda_T$ (= $C_{w,o}Re_\phi^{-0.2}$), where λ_T is the turbulent flow parameter that controls the flow structure in many rotor-stator systems [19]. As described in the introduction, there is complex interaction and mixing between ingress fluid from the mainstream annulus and the egress flow from the wheel-space in the region of the rim seal, and this affects the swirl ratio of the ingested fluid entering the wheel-space. Following Wang and Wilson [17], the inlet boundary value of swirl ratio for the ingress flow imposed as a uniform inflow at the simplified rim seal inlet at the outer radius shown in Fig. 3 is therefore estimated here using the following empirical mass-flow weighted average:

$$\beta_i = \frac{\dot{m}_i\beta_a + \dot{m}_o\beta_o}{\dot{m}_i + \dot{m}_o} \tag{3}$$

For the University of Bath experimental rig, the swirl ratio of flow in the annulus (in the seal clearance region downstream of angled stationary vanes) is $\beta_a \approx 1.8$, and β_o is taken to be zero. The value of $\dot{m}_i$ is deduced from the value of $\Phi i/\Phi_o$ shown in Fig. 2 for any particular chosen value of Φ_o. (The same mass-weighted averaging method is used to estimate the corresponding concentration of tracer gas, introduced as a passive scalar entering with the sealing flow, in the mixed-out fluid entering the wheel-space as ingress. The resulting computed distributions of concentration-based sealing effectiveness were presented and described by Wang and Wilson [17].)

For boundary conditions for the heat transfer computations a uniform surface temperature (20° C) was used for the rotor and other surfaces were assumed adiabatic. The temperature of the sealing flow at inlet was 57.5° C as in the experiments. The air temperature in the mainstream annulus in the experiments was 14.5° C. Using this as the temperature for the prescribed ingestion resulted in poor agreement with measured heat transfer, indicating that the mixing between annulus fluid (ingress) and wheel-space fluid (egress) in the seal region also affects the temperature of the ingested flow entering the wheel-space. A mass-weighted averaging as described above was therefore also adopted as follows for the estimated temperature T_i of the ingested fluid at inlet:

$$T_i = \frac{\dot{m}_i T_a + \dot{m}_o T_o}{\dot{m}_i + \dot{m}_o} \tag{4}$$

Here T_a and T_o are the air temperature in the annulus (14.5° C) and the temperature of the sealing flow (57.5° C) respectively.

4. HEAT TRANSFER RESULTS

Computations were carried out for different values of sealing flow rate in order to find the concentration and thermal (adiabatic) sealing effectiveness in each case. The test conditions correspond to individual selected cases chosen from those shown in Fig. 2 for an axial-clearance seal geometry for particular values of the sealing flow parameter Φ_o.(corresponding to roughly equal values of the turbulent flow parameter λ_T as described above.) The values of the flow parameter or λ_T considered are given in subsequent figures.

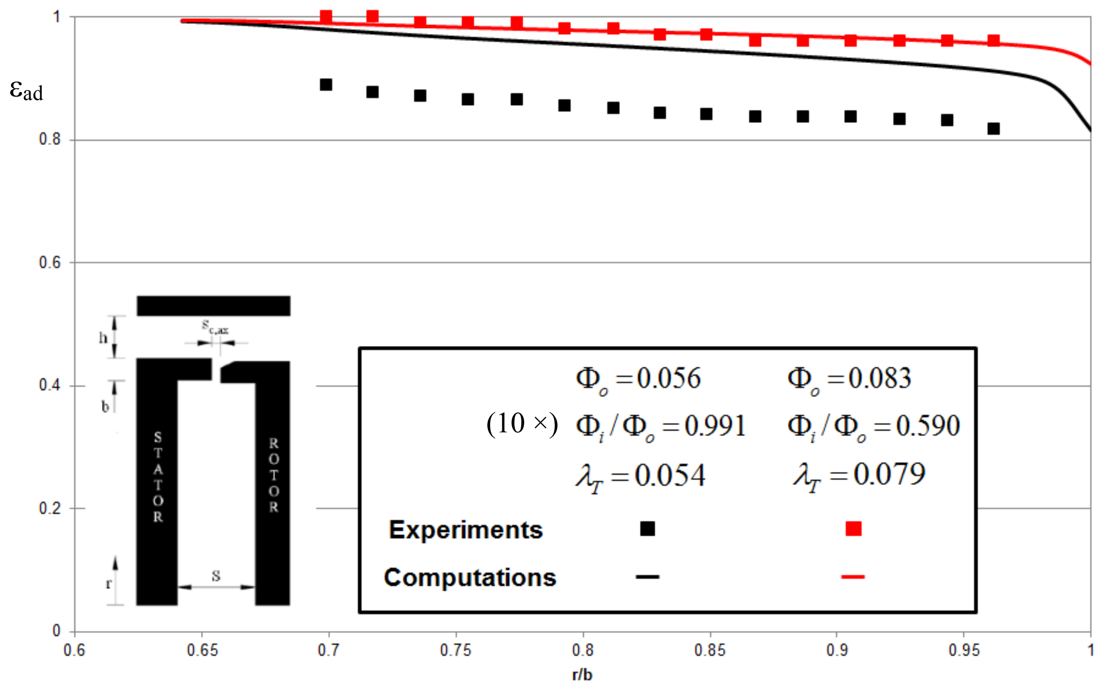

Fig. 4 Computed radial distribution of adiabatic effectiveness on the rotor compared with experiments [1]

The adiabatic sealing effectiveness is given by [1]:

$$\varepsilon_{ad} = \frac{T_{ad} - T_a}{T^*_{ad} - T_a} \tag{5}$$

Here T_{ad}* is the value of T_{ad} when there is no ingress and T_a is the temperature of the air in the annulus. Fig. 4 shows computed radial distributions of ε_{ad} on the rotor and comparison with the measurements of Pountney et al [1] for two different values of the sealing flow parameter Φ_o. The local sealing effectiveness on the rotor decreases as non-dimensional radius r/b increases, the explanation for this being that ingested fluid, which is drawn into the stator boundary layer at high radius (see Fig. 1), is subsequently entrained into the rotor boundary layer as fluid migrates axially from the stator to the rotor. Fig. 4 shows that the computed adiabatic effectiveness over-predicts the measurements for the lower of the two sealing flow rates considered (Φ_o = 0.056), which corresponds to the case having the higher amount of ingestion. The adiabatic and concentration-based definitions of effectiveness gave almost identical computed distributions on the rotor at each of the two sealing flow rates.

Nusselt numbers at the rotor surface were calculated from:

$$Nu = \frac{qr}{k(T_s - T_{ad})} \tag{6}$$

Here q is the heat flux to the rotor, T_s is the rotor surface temperature and T_{ad} is the adiabatic surface temperature. For the computations, T_{ad} was obtained for each case considered from a separate computation at the same conditions of sealing flow rate and ingested flow rate but using an adiabatic rotor wall. The experimental data were obtained from the thermochromic liquid crystal (TLC) data and the solution of the 1D Fourier equation for heat transfer [22].

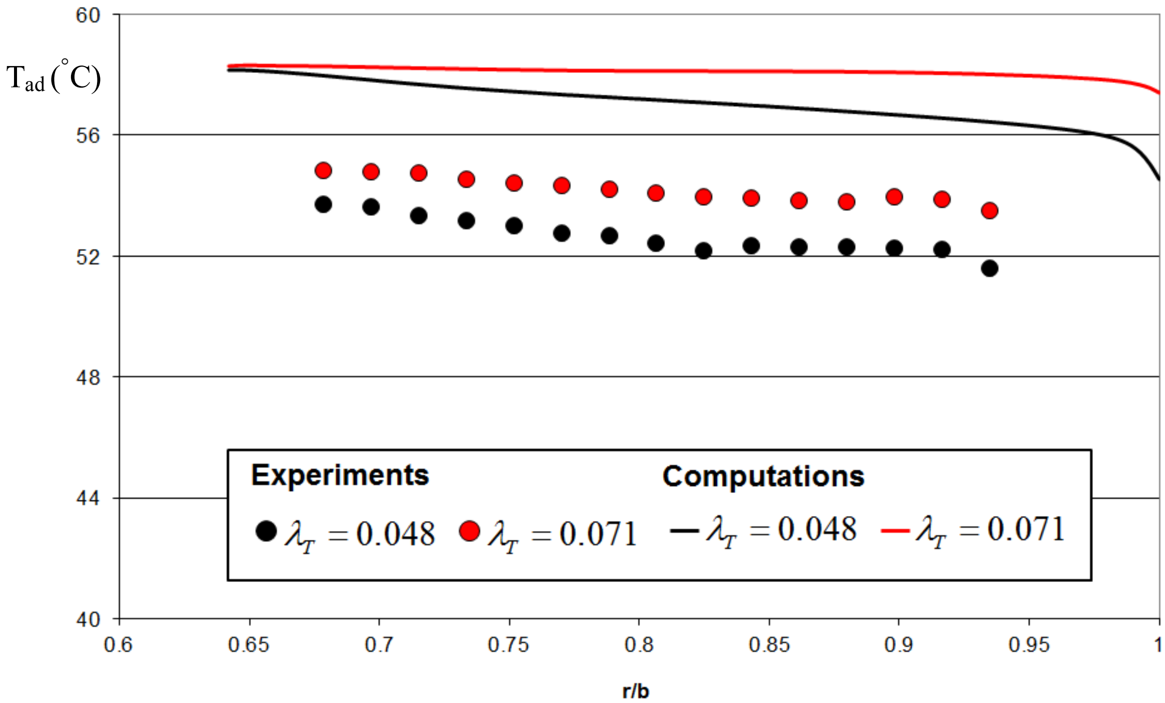

Fig. 5 Computed rotor adiabatic disc temperature T_{ad} compared with experiments [22]

Fig. 5 shows computed and measured values of adiabatic surface temperature for two cases. The adiabatic computations produce reasonable values of T_{ad} and show similar trends to the experiments, although the computed T_{ad} are higher than the measured values.

Fig. 6 shows the computed radial variation of $NuRe_{\phi}^{-0.8}$ (as used by Pountney et al [1]) on the rotor in comparison with the experimental results [1]. (The method for calculating the uncertainties in the experimental results as shown in Fig. 6 is described in [1].) Both the computations and the experiments show that, away from the inner and outer radii, Nu is almost invariant with radius. The computed results capture qualitatively some features of the experimental results, but over-predict the measurements. The main reason for the differences is the over-prediction of adiabatic surface temperature as shown in Fig. 5. Fig. 6 shows that Nu decreases as sealing flow rate λ_T decreases (so that the amount of ingestion increases). The higher swirl associated with higher ingestion flow rates (as is represented in Eq. (3)) raises the swirl ratio in the fluid core in the wheel-space, and this acts to reduce heat transfer to the rotor. This effect is greater in the computations than in the measurements. Further computations showed that the computed Nusselt numbers were not sensitive to variation of the estimated boundary conditions for the ingress flow swirl ratio and temperature.

In order to study the effect of stator heat transfer, further computations were carried out with the stator at the same temperature (20° C) as the rotor. Fig. 6 shows the effect of the stator heat transfer on the radial variation of Nu on the rotor. The heat transfer to the stator reduces Nu, but the effect is small. This is a similar finding to that of Javiya et al. [23] from a different study of a pre-swirl rotor-stator system.

The temperature of the rotating core of fluid in the wheel-space is also an important factor in controlling heat transfer to the rotor. Fig. 7 shows the radial variation of non-dimensional core temperature, θ_∞, defined by:

$$\theta_\infty = \frac{T_\infty - T_{in}}{T_o - T_{in}} \tag{7}$$

Here T_∞ is the air temperature at an axial distance z/s = 0.3 from the stator; this location is considered to be in the fluid core outside the boundary layer on the stator. T_{in} is the initial temperature of the air in the transient

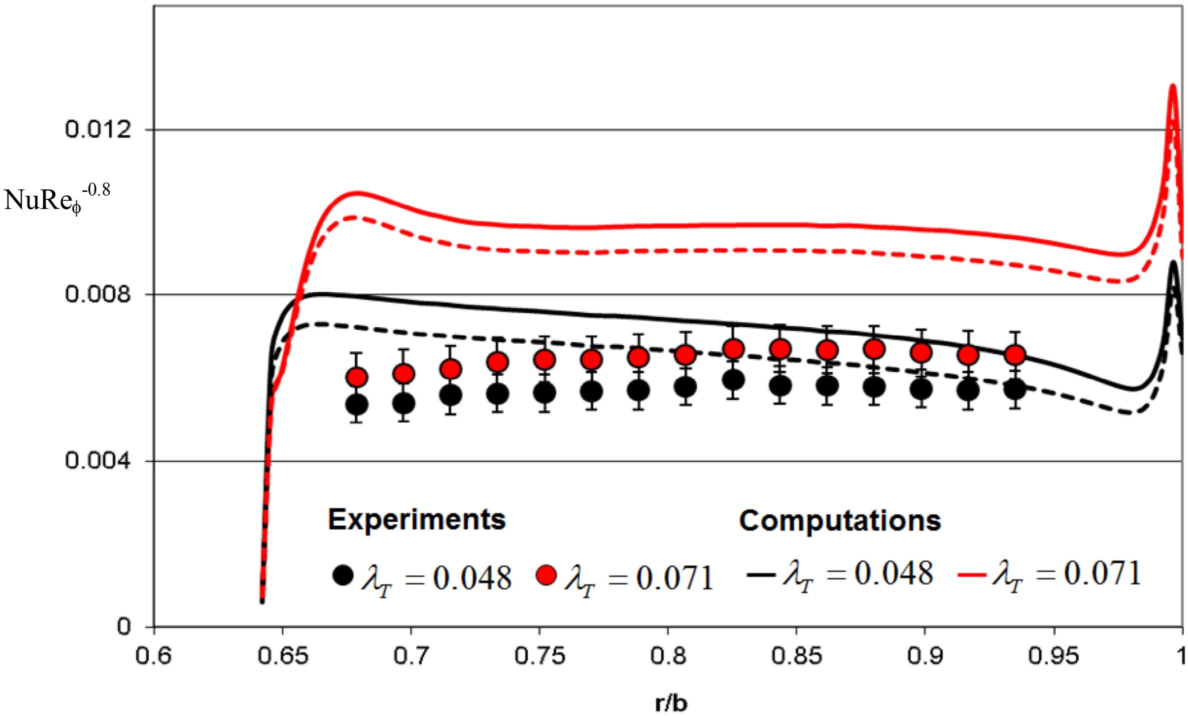

Fig. 6 Computed radial variation of Nusselt number for $Re_\phi = 8.17 \times 10^5$ compared with experiments [1] (solid lines – adiabatic stator , dashed lines – fixed temperature stator)

experiment before heating was applied to the sealing flow to introduce a step-change in its temperature to T_o. In the present computations, the temperature of the rotor (20° C) is used as T_{in}.

Fig. 7 shows that the cold ingested air reduces the core temperature at high radius for both the computations and the experimental results. The computed core temperatures are higher than the measured values. It should be noted that the core temperatures will also be affected by heat transfer to the stator and that the thermal conditions on the stator for the computations and the experiments are likely to be different. Despite this, the measured variation of θ_∞ is predicted reasonably well by the computations.

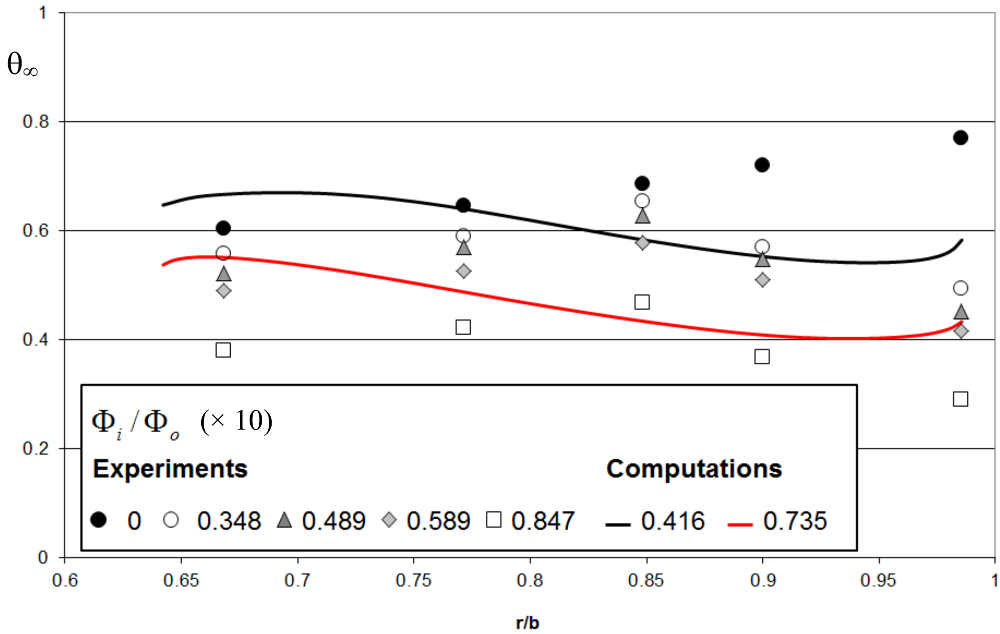

Fig. 7 Computed radial variation of non-dimensional core temperature compared with experiments [22]

5. CONCLUSIONS

A simplified steady flow axisymmetric computational fluid dynamics model has been used to compare predicted heat transfer in a rotor-stator system including the effects of ingestion with previously published measurements made using a transient experimental technique. The mas-flow rate of ingested fluid is deduced from experimental measurements of ingestion and estimated incoming values of swirl ratio and temperature are prescribed. The simplified model matches qualitatively features of the experimental measurements for the heat transfer on the rotating disc, and for the temperature in the rotating core of fluid in the wheel-space between the boundary layers on the rotor and stator discs. The computed Nusselt numbers capture qualitatively some features of the experimental results, but over-predict the measurements.

ACKNOWLEDGEMENT

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NOMENCLATURE

a	wheel-space inner radius	(m)	β	swirl ratio	(-)
b	wheel-space outer radius	(m)	ε	sealing effectiveness	(-)
C_W	non-dimensional flow rate	(-)	λ_T	turbulent flow parameter	(-)
G	wheel-space gap ratio (=s/b)	(-)	Ω	disc rotation rate	(s^{-1})
G_c	seal gap ratio (s_c=/b)	(-)	θ_∞	non-dimensional core temperature	(-)
Nu	Nusselt number	(-)	Φ	sealing flow parameter	(-)
r	radius	(m)			
Re_ϕ	rotational Reynolds number	(-)			
s	wheel-space axial gap	(m)			
s_c	axial seal clearance	(m)			
y+	non-dimensional wall-distance	(-)			
z	axial distance from stator	(m)			

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